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ELASTIC-PLASTIC ANALYSIS OF SCREW THREADS

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An elastic-plastic analysis method is suggested for screw thread teeth. In this method a single tooth is analyzed using boundary conditions to simulate a long chain of identical teeth. A set of five different loads are suggested to simulate pressure and shear on each flank along with a general stress field in the component. An example is worked out for a British Standard Buttress thread form. Data is presented from the example to show that friction is a very important parameter.

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INTRODUCTION

The problem of stress concentrations in screw threads has long been obscured by the larger number of parameters involved and the lack of a systematic approach which could help to explain the variation that is in any experimental program. The object of this work is to try to cut through those problems and try to present a useful, organized approach which can encourage more work in this general area.

An example of the large number of parameters is the geometry description shown in Figure 1. While these dimensions may be of use to the designer to insure that the component will fit together, the stress analyst needs only a few of them. The major geometry parameters are the primary flank angle (α), secondary flank angle (β), and root radius (R). The primary loading parameters are the applied load (W), its angle (γ), and position (b). These last three parameters follow the convention of Heywood.[1] A further simplification is to nondimensionalize all linear dimensions to the pitch (P).

The very high performance requirements of military hardware have in the past produced a new thread form[2,3] for use on cannon breech components. During the development of the Watervliet Buttress thread, the Heywood equation[1]

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[1]R. B. Heywood, "Tensile Fillet Stresses in Loaded Projections," Proceedings of the Institute of Mechanical Engineering, Vol. 160, p. 124, 1949.

[2]R. E. Weigle, R. R. Lasselle and J. P. Purtell, "Experimental Investigation of the Fatigue Behavior of Thread-Type Projections," Experimental Mechanics, Number 5, Vol. 3, pp. 105-111, 1963.

[3]R. E. Weigle and R. R. Lasselle, "Experimental Techniques for Predicting Fatigue Failure of Cannon-Breech Mechanisms," Experimental Mechanics, February 1965.

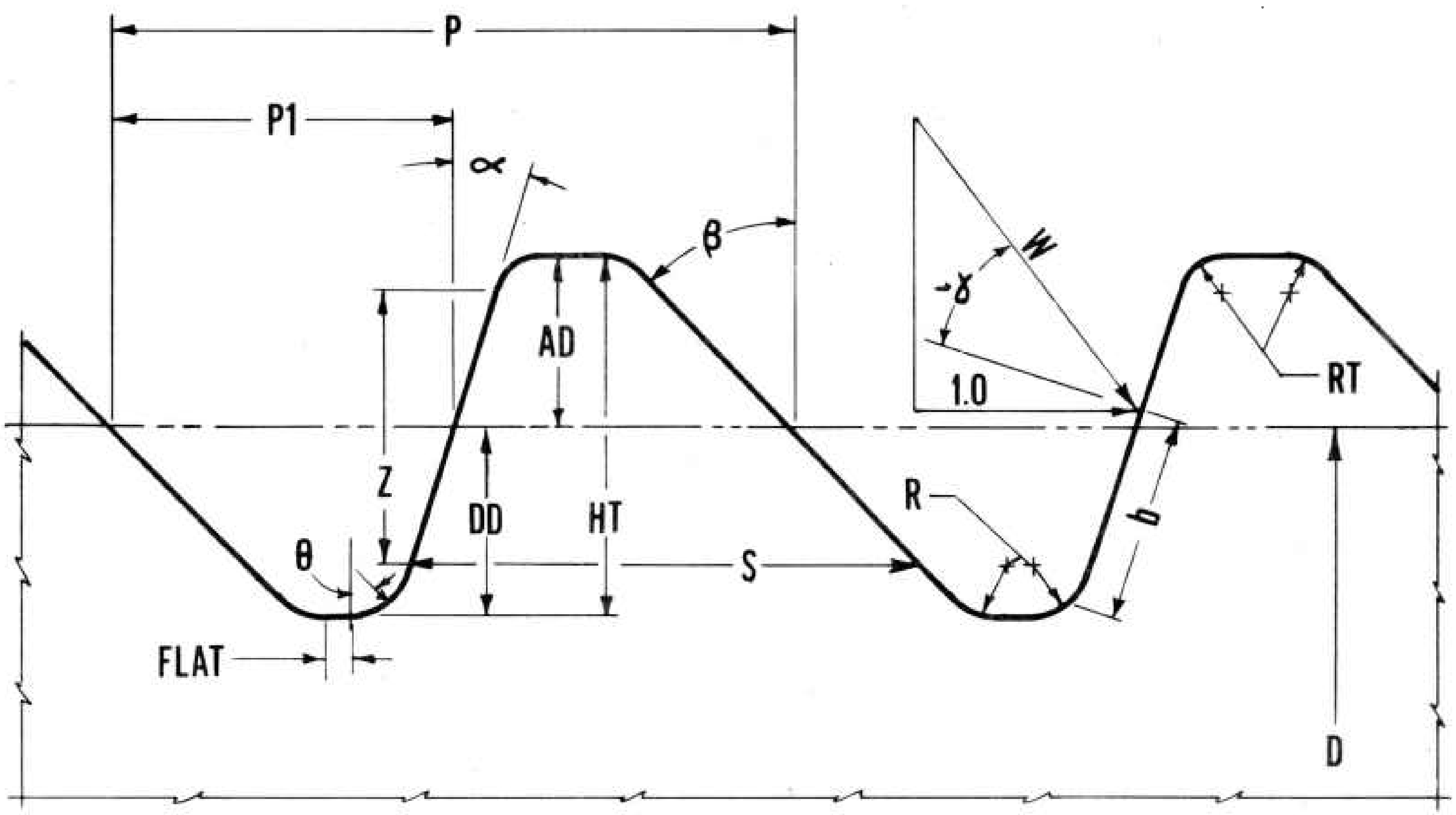

Figure 1. **THREAD GEOMETRY AND LOAD PARAMETERS**

was used to choose geometry parameters for testing with good success. The Heywood stresses, however, were never correlated with test results. The reason for this was pointed out by this author[4] and it is simply that Heywood isolated his teeth so that only effects due to the load on the teeth would be present. In most experiments, the stress in the fillet of a thread is the result of load on the thread plus a stress concentration of the general stress field in the component.

In a recent paper[5] this author offered an elastic stress concentration approach to screw threads. In this work the overall loading on a thread is resolved into two forces parallel and normal to the pitch line. These are divided by the area on the pitch surface to produce two average stresses, radial stress and shear transfer rate. Of these, shear transfer rate is used to normalize all stresses, and the radial stress is used in a plot with the maximum fillet stress to produce a curve which is a characteristic of a particular thread form. This curve is usually generated as the coefficient of friction is varied from -1.0 to 1.0. where the sign denotes the direction of the friction vector, positive being away from the fillet. This sign convention gives the radial stress the same sign convention as all other stresses with tension positive. With this method an axial body stress in terms of a uniform remote tension can be easily added to produce a family of characteristic curves.

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[4] G. P. O'Hara, "Finite Element Analysis of Treaded Connections," Proceedings of the Army Symposium on Solid Mechanics, AMMRC, MS-74-8, pp. 99-119, 1974.

[5] G. P. O'Hara, "Stress Concentration in Screw Threads," ARRADCOM Technical Report, ARLCB-TR-80010, 1980.

The above work is all elastic and certainly only looks at less than half of the overall problem. Elastic-plastic analysis adds a new set of problems to the analysis. First is that it is possible to identify five different plastic zones in a single tooth (Fig. 2), the axial stress zone, the Heywood zone, the secondary flank zone, the shear failure zone, and the bearing failure zone. It is difficult to imagine a problem in which only one of these is present and the usual case is where plasticity involves three or more of those zones working together with each starting at its own load.

The major factor that complicates elastic plastic analysis is that it is directly linked with the material stress-strain curve, and a general solution can be found only for materials with similarly shaped curves. For this report the assumed material will be 7075-T6 aluminum (Fig. 3)[6] with a proportional limit of 65 Ksi and 0.2 percent offset yielded of 72 Ksi. This is an engineering stress-strain curve defined out to 6% strain.

ELASTIC PLASTIC METHOD

The NASTRAN Rigid Format 6, Piecewise Linear Analysis is covered in the Theoretical Manual[7] and uses the triangular ring element (CTRIARG) which was implemented in a parallel program with the trapezoidal element reported by Chen[8]. In this method the number and size of the linear steps is selected by the user before the run. It is the duty of the user to select steps which

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[6] "Aero Space Structural Metals Handbook," Mechanical Properties Data Center, Belfour Sluton Inc., Code 3207, pp. 15-16.

[7] "The NASTRAN Theoretical Manual," R. H. MacNeal, Editor, NASA SP-221, Level 15.

[8] P. Chen, G. P. O'Hara, "Implementation of a Trapezoidal Ring Element in NASTRAN for Elastic-Plastic Analysis," ARRADCOM Technical Report, ARLCB-TR-79034, December 1979.

PLASTIC ZONES

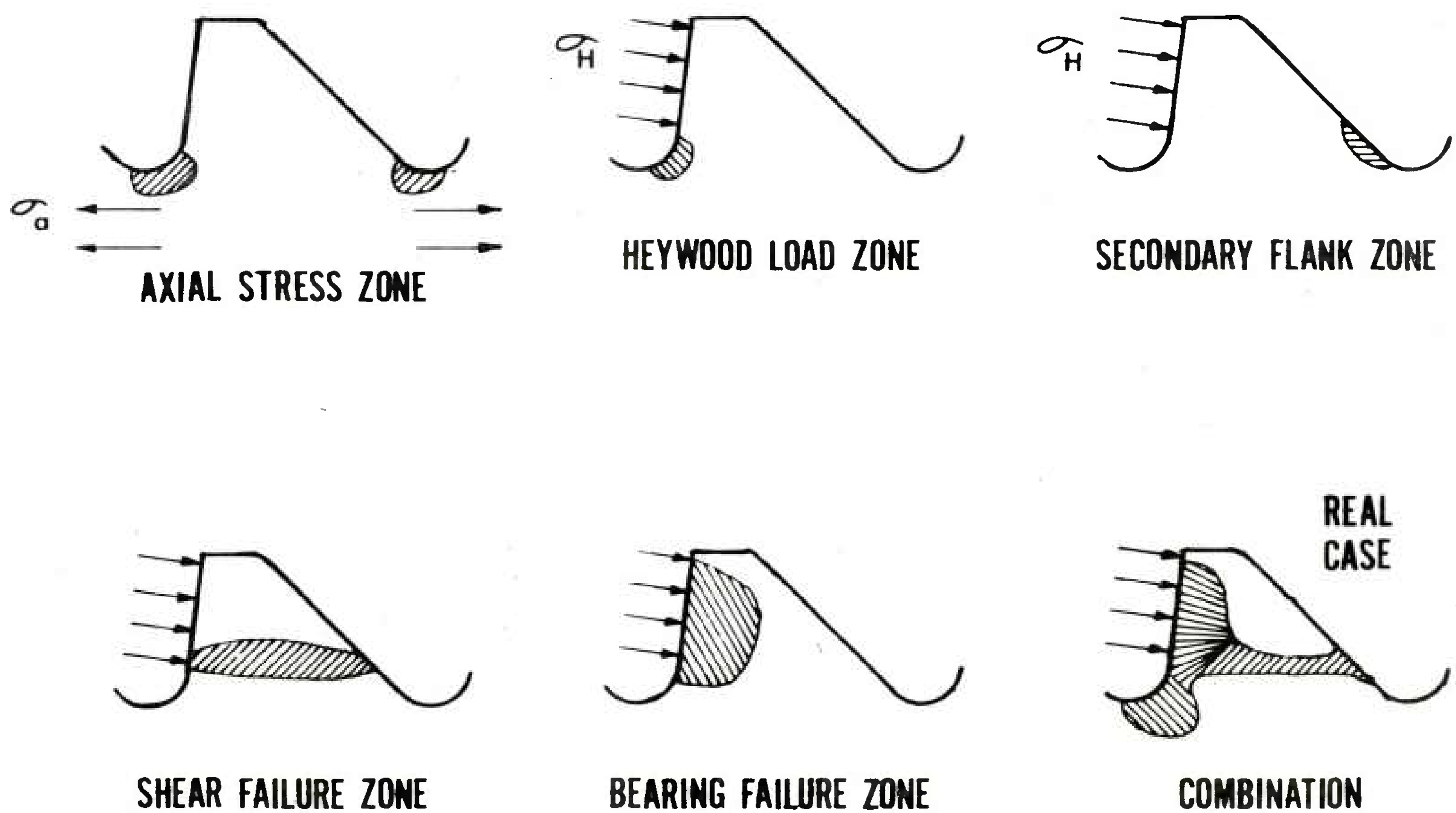


FIGURE 2

STRESS VS. STRAIN

FOR 7075-T6
ALIMINIUM

5.6% ZN.

2.5% MG.

1.6% CU.

0.3% CR.

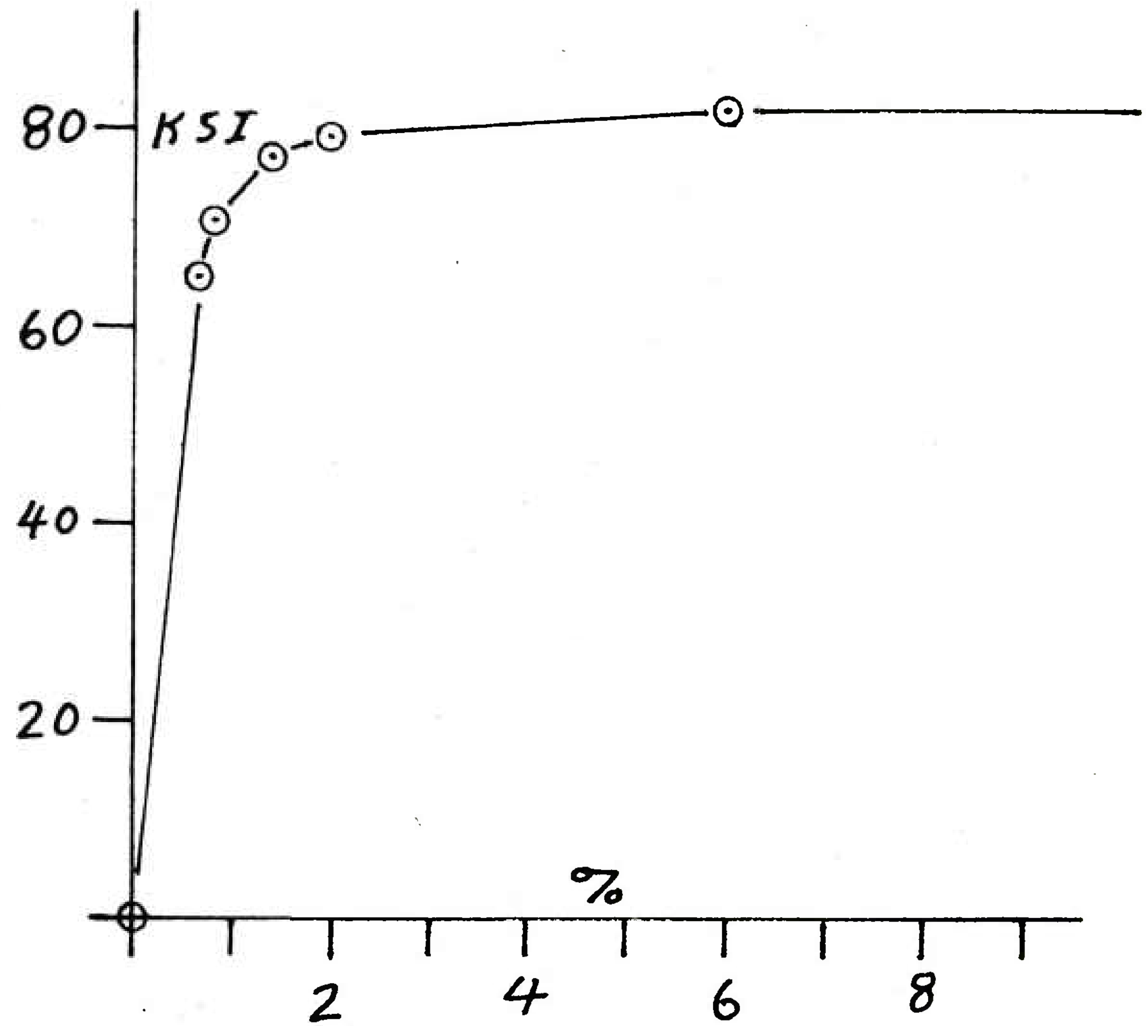


FIGURE 3

produce adequate results within the limitations of the available computer time. The program then selects the slope off the stress-strain curve by extrapolating the change in effective strain for the current load step out to the end of the next load step and using an estimated elastic modulus (E)

$$E_i = \frac{\sigma_{i+1} - \sigma_i}{e_{i+1} - e_i} \tag{1}$$

Where σ_{i+1} and e_{i+1} are estimated values. This will be equal to the slope of one of the liner segments of the input stress-strain curve only when both points fall within the same liner segment of the curve. In the case of a zero modulus the element is assumed to have no increase in stiffness and a zero element stiffness matrix is generated.

The use of the stepped constraint input is not normally allowed because of the ambiguity that would exist if both forces and constraints were stepped together. This can be overcome using a small DMAP alter package in the executive control deck when only constraint input is to be used. Under these conditions it would seem that superior results could be expected because the extrapolation is done on the basis of strain.

The solutions in this report have been set up on the basis of 13 load steps, however, in one case the solution was truncated when a portion of the structure exceeded the defined stress strain curve and entered the zero slope region. When this has happened to all elements connecting any grid point a singular body stiffness results and the solution is stopped. This results when the modulus (E) is zero and the element stiffness matrix becomes zero.

BOUNDARY CONDITIONS

In this work a small finite element grid (Fig. 4) will be used to simi- late the behavior of a long chain of identical threads. This requires boundary conditions for the three surfaces where the model is cut out of the larger problem as well as applied loads. These surfaces are the two radial planes and an axial cylinder. These surfaces will be treated differently for axial load and the Heywood loads on the thread bearing surfaces.

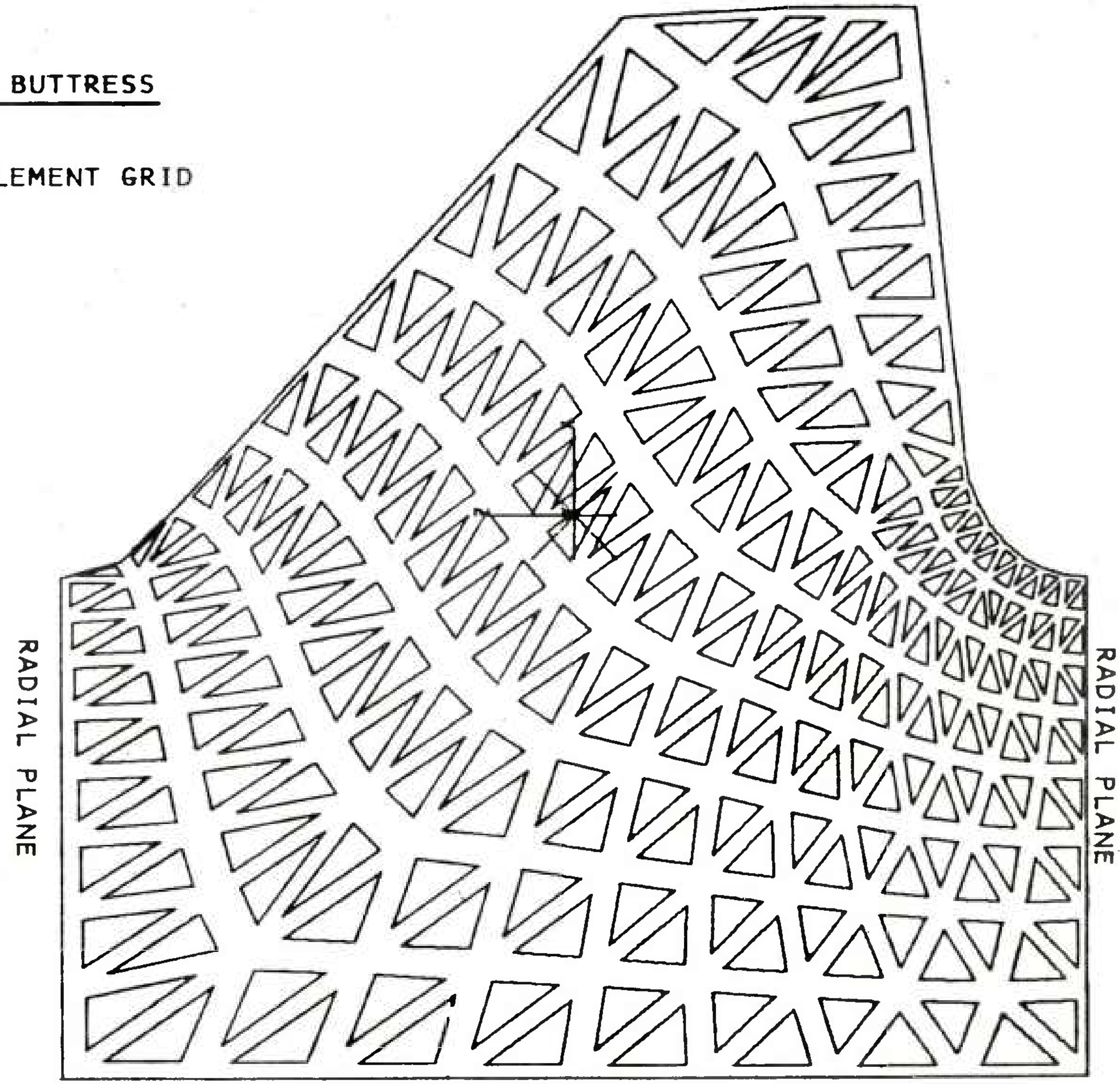
The grid points on the axial cylinder must be constrained to replace the bulk of the body material. For the axial stress input these points are free in the axial direction and are constrained to a fixed displacement in the radial direction. This radial displacement accounts for Poisson contraction in the body. The grid points in the radial planes are generated at the same radial locations to allow them to be constrained in pairs, one in each radial plane. The radial displacement of each point of a pair is equal and the relative axial displacement of all pairs is the same. This forces the radial plane to conform to the same deformed shape while being free to distort out of the planes. In the elastic-plastic solution for axial stresses the constraint values for Poisson's constraint and relative elongation are stepped together to produce a piecewise constraint input condition.

In the solution for Heywood loads the object is to react the load out in shear, therefore the grid points on the axial cylinder are given a zero displacement in the axial direction. This zero displacement is also given to the radial displacements to simulate a stiff structure. The two radial planes

BRITISH STANDARD BUTTRESS

BASIC FINITE ELEMENT GRID

FIGURE 4



retain the same constraints as for the axial loads, however, the relative axial displacement is set to zero. In this case forces on the bearing surface are stepped to produce the piecewise loads.

Figure 5 shows the forces on a particular thread tooth. There is a primary and a secondary bearing flank with a pressure and a shear load on each. The primary flank is the one which is intended for force transfer. The secondary flank becomes loaded under reverse loading or when displacement removes the radial clearance. In this paper only uniform loads on the primary flank will be used. In this case the pressure and shear loads are added into an overall load W which is then resolved into a radial load (L_R) and axial load (L_a). These are the loads which are averaged over the area at the pitch line to produce the radial stress (σ_r) and the shear transfer rate (τ_R).

EXAMPLES OF ELASTIC-PLASTIC ANALYSIS

In this report four examples of elastic-plastic analysis will be shown for the thread form used, the British Standard Buttress. This form appears as a high strength thread in several Army structures such as the M68 cannon breech and some kinetic energy armor piercing projectiles where it seems to have been selected because of the low radial load component. The loading are all uniform applied loads and include one axial stress load and three Heywood type loads. The finite element grid is shown in Figure 4 with the element shrunken to expose each side. This grid has a pitch diameter of 10.0 times the pitch length.

LOADS

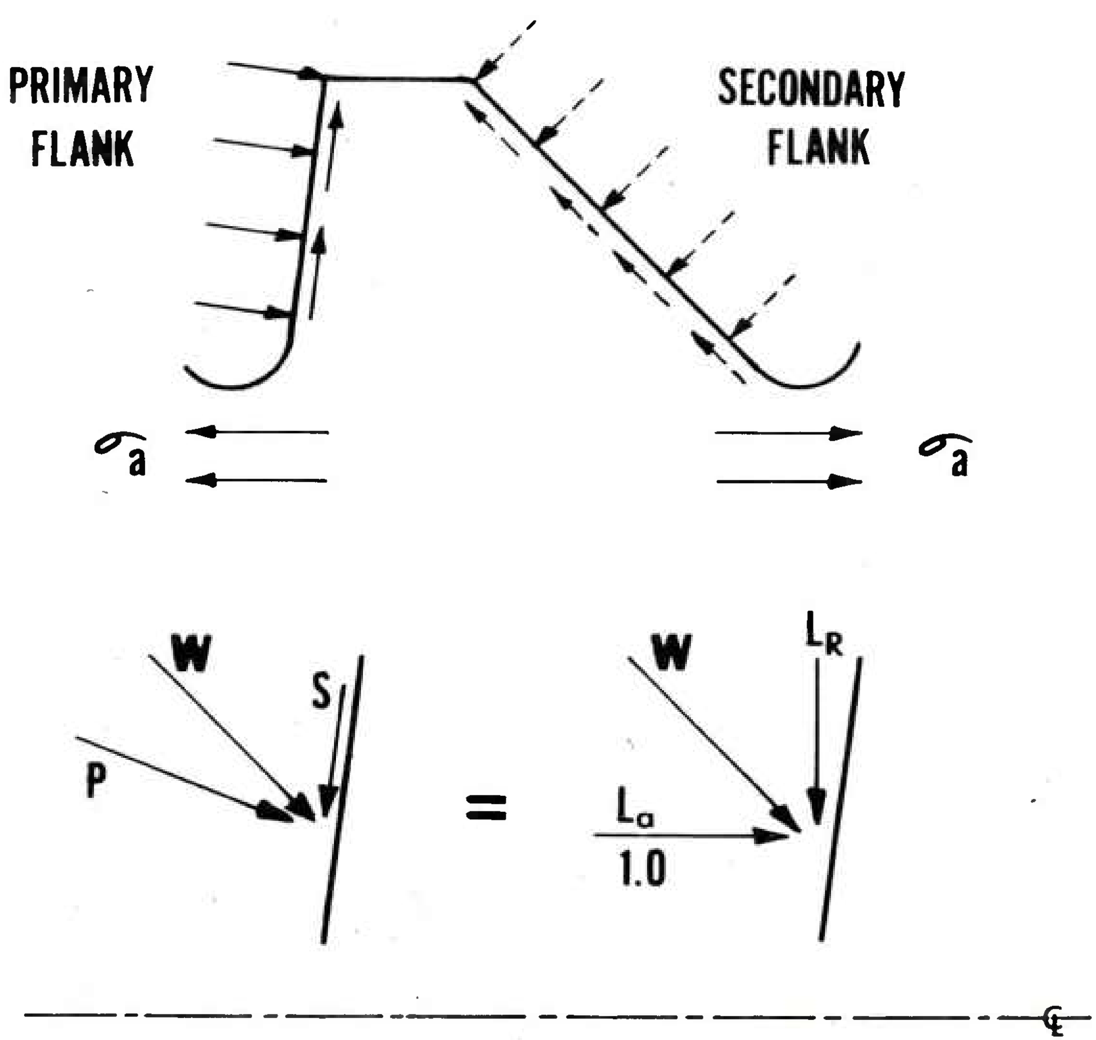


FIGURE 5

The first load is an axial stress in the body of the component with a peak of 65 Ksi. This is done by constraining the relative axial displacement of the radial planes to a fixed value and stepping that value in the piecewise solution. The axial cylinder is stepped in a similar way to produce the Poisson contraction. Figure 6 shows a shrunken element plot of those elements which have become nonlinear. In this plot all the elements shown are above the proportional limit stress of 65 Ksi. The elements shown doubled are above the conventional yield stress at the .2% offset point.

The three Heywood loads use three different values of friction coefficients -.5, 0.0, and +.5 where the sign on friction denotes the direction of the friction vector. Figures 7, 8, and 9 show the plots of nonlinear elements. It should be noted that the shear transfer rate for Figure 9 is lower than the other two. This is because that solution exceeded the 6% strain maximum of the stress-strain definition and the solution was stopped at that point. The arrows in these plots point out the element where the maximum stress occurs which is different in each of these solutions and the axial stress plot.

These plots of nonlinear elements show one part of the overall picture. The next thing to look at is the maximum stress in the fillet. Figure 10 is the curve of fillet stress vs axial stress for the axial load. This solution was stopped at this point because the constraint input leaves some question about the nominal input stress when the bulk stress exceeds the yield point. The maximum fillet stresses for all three Heywood loads are plotted against shear transfer rate in Figure 11. The very high values for the stresses are

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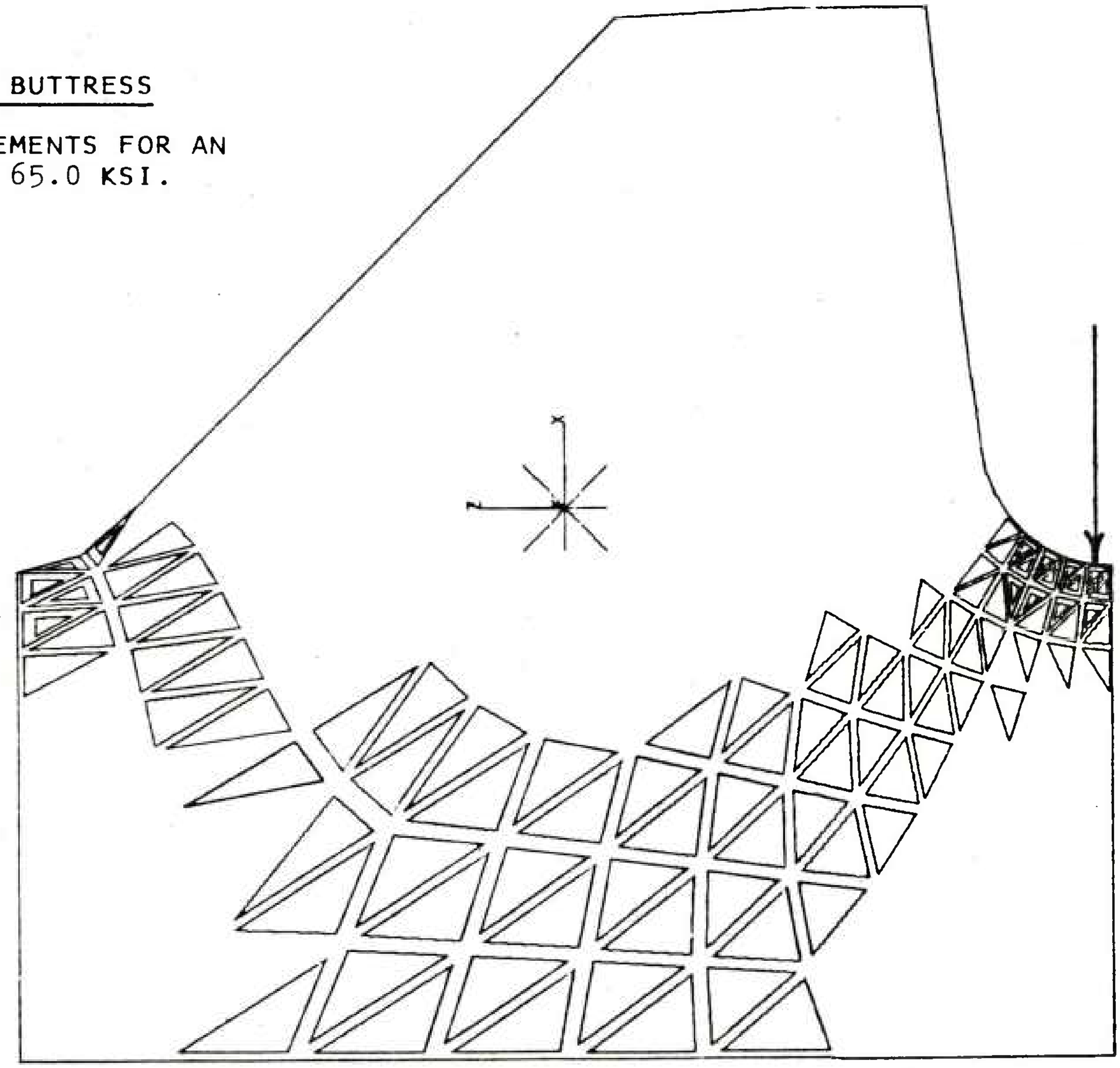
NON-LINEAR ELEMENTS FOR AN
AXIAL LOAD OF 65.0 KSI.

FIGURE 6



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NON-LINEAR ELEMENTS FOR A
HEYWOOD LOAD WITH
FRICTION = -.5 AND
SHEAR TRANSFER = 31.0 KSI.

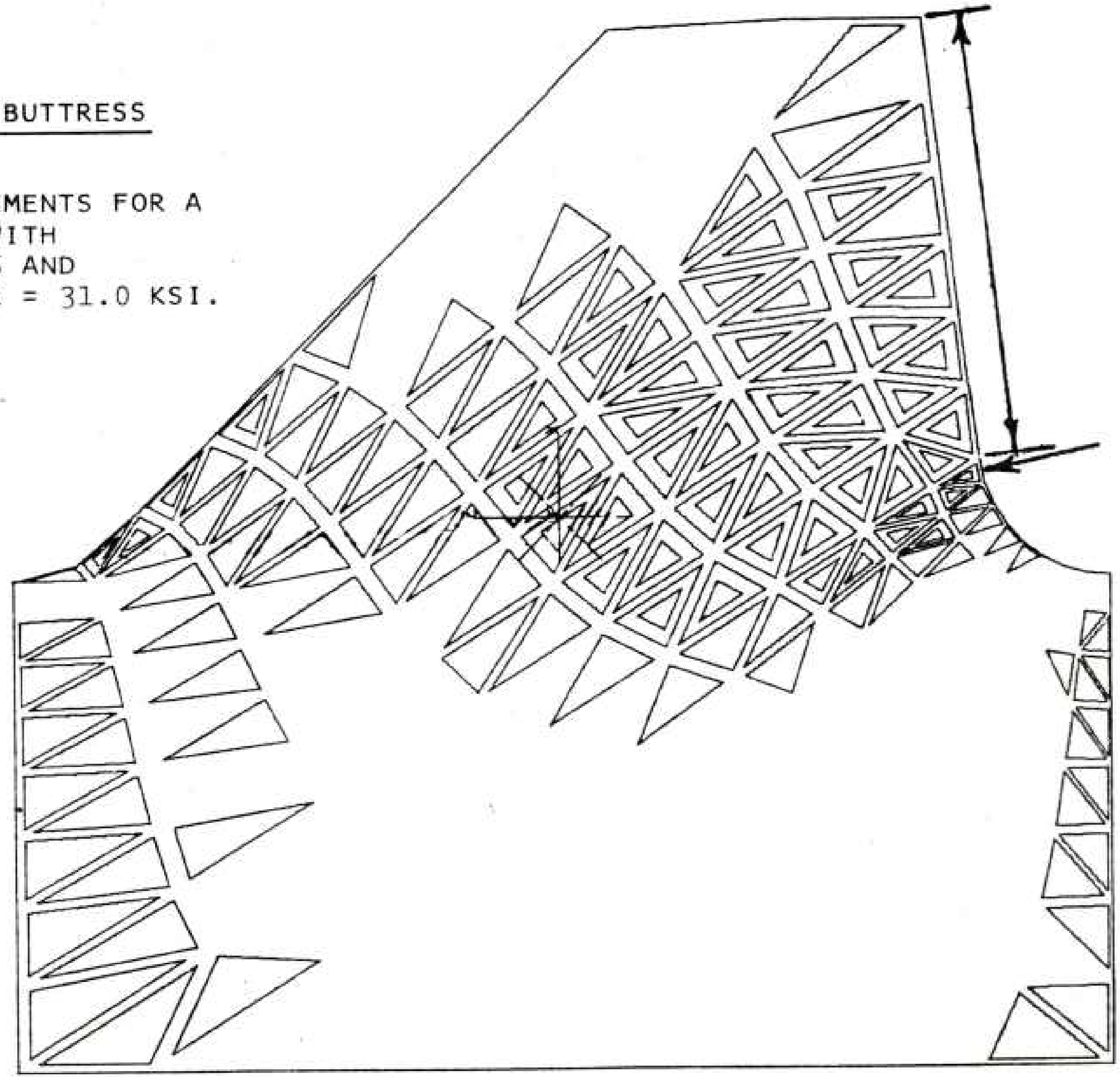

FIGURE 7

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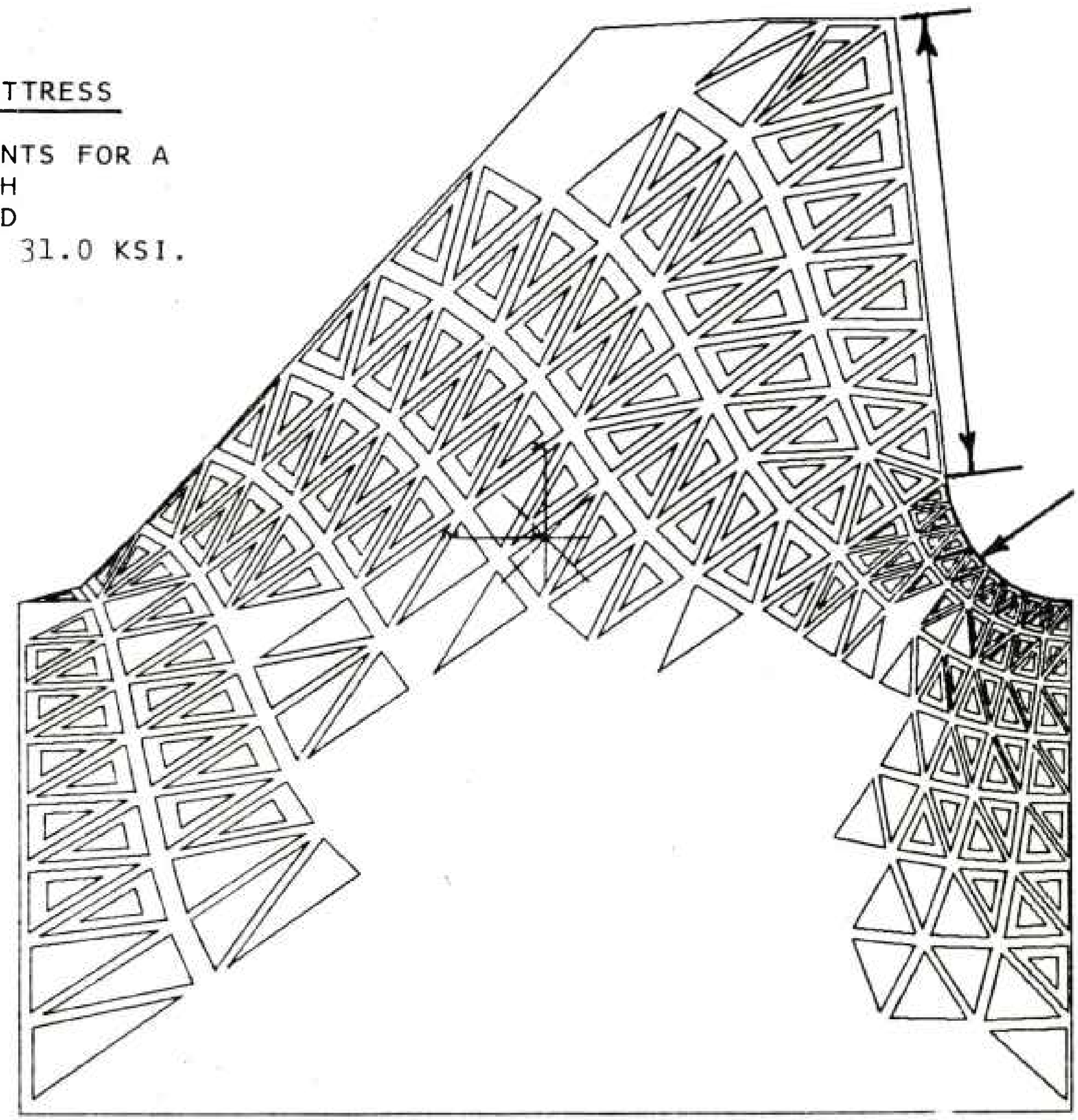
NON-LINEAR ELEMENTS FOR A
HEYWOOD LOAD WITH
FRICTION =0.0 AND
SHEAR TRANSFER = 31.0 KSI.

FIGURE 8



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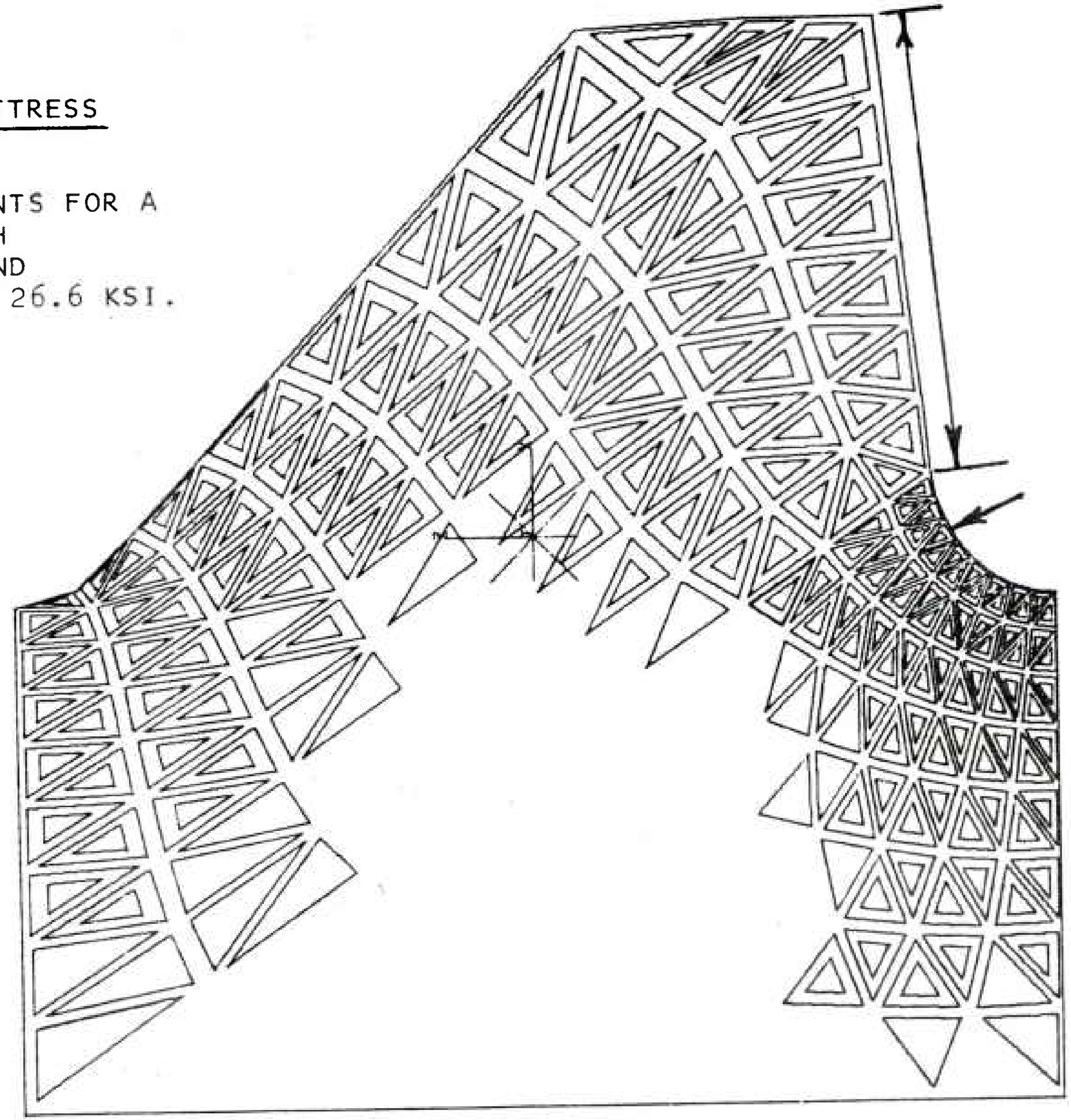
NON-LINEAR ELEMENTS FOR A HEYWOOD LOAD WITH FRICTION = +.5 AND SHEAR TRANSFER = 26.6 KSI.

FIGURE 9



FILLET STRESS
VS.
AXIAL STRESS

FOR THE

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BUTTRESS

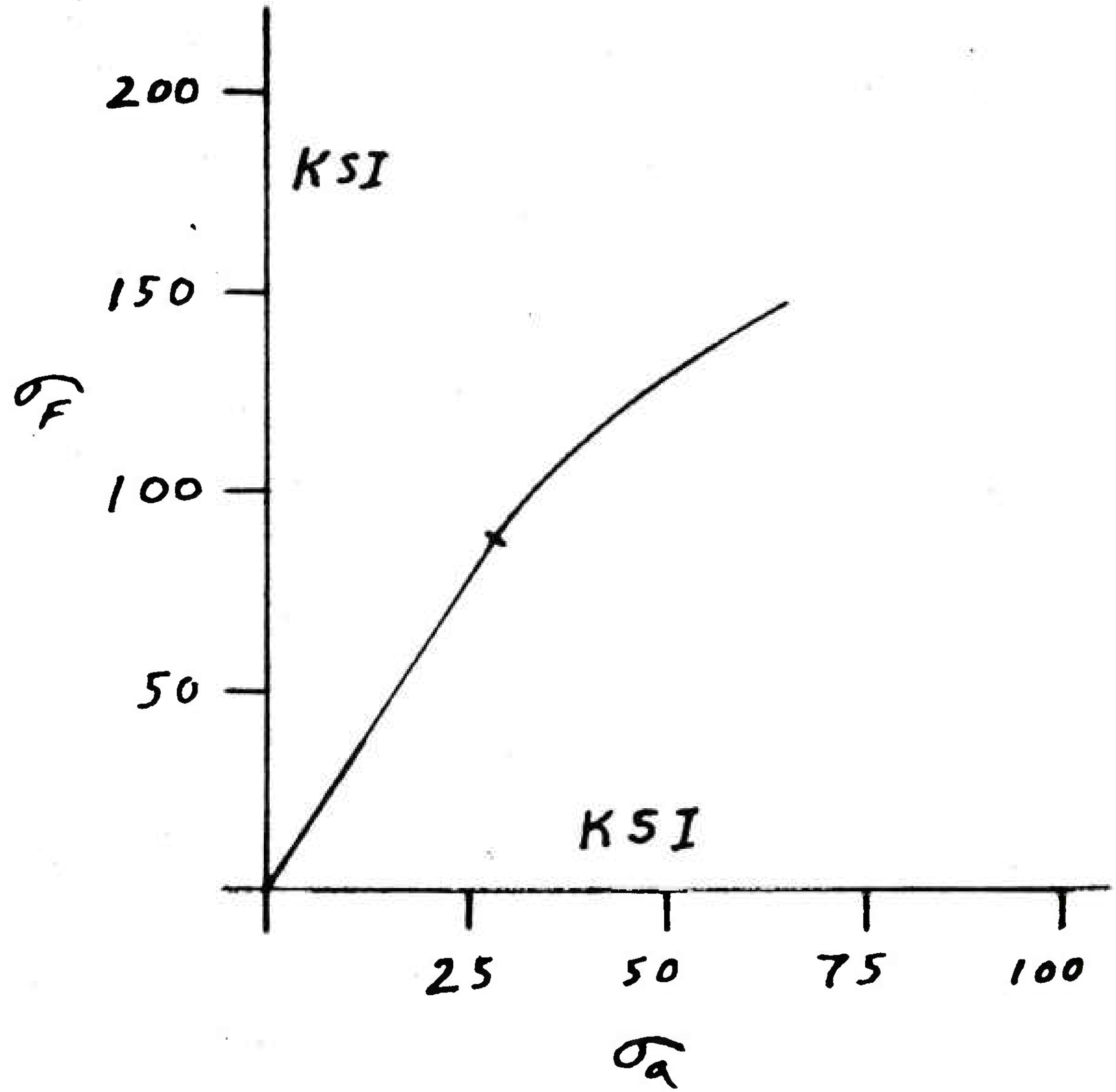

FIGURE 10

FILLET STRESS
VS.
SHEAR TRANSFER

FOR THE

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BUTTRESS

USING THREE
DIFFERENT
FRICTION
VALUES

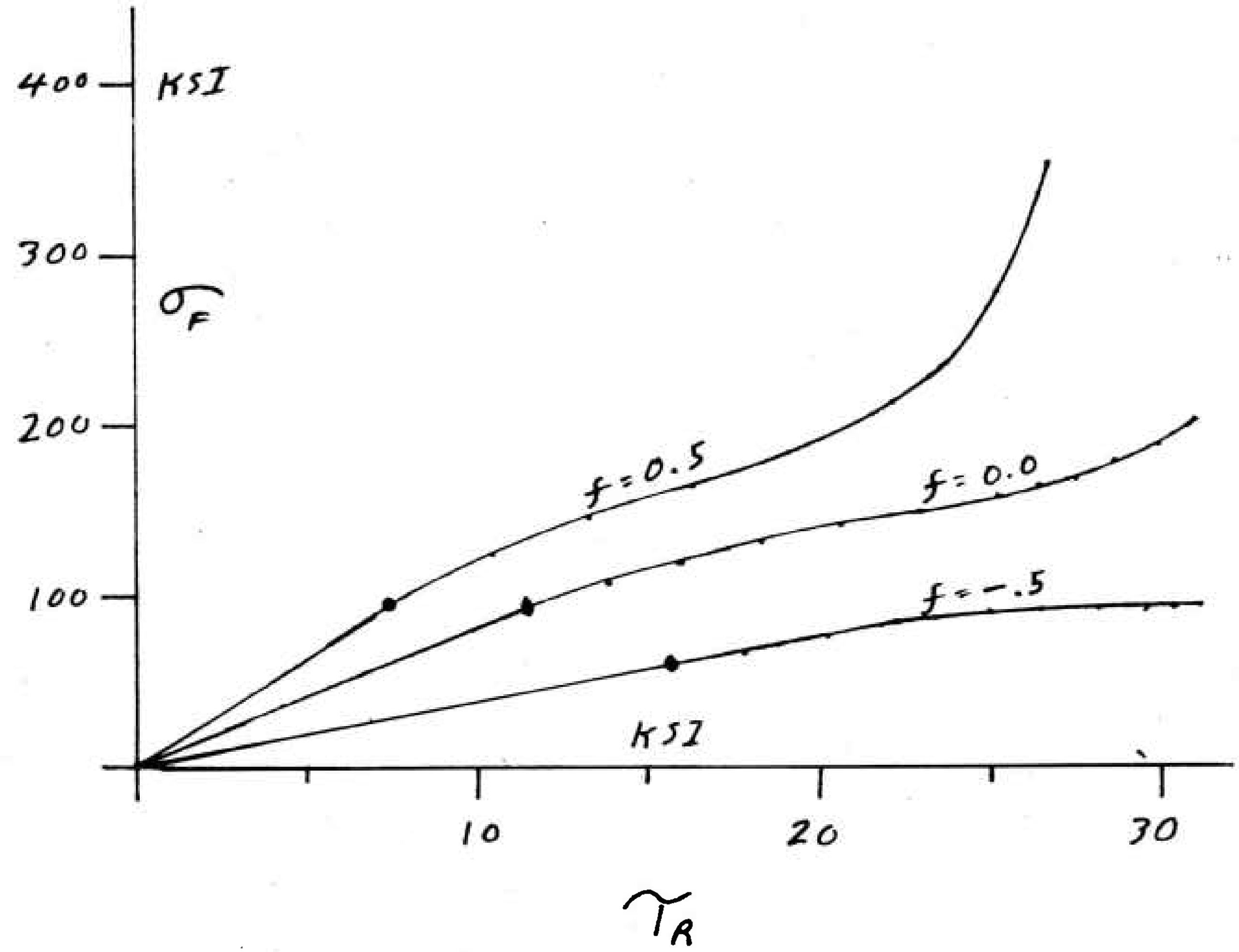

FIGURE 11

the result of the multi-axial stress state in the fillet and other than that the plot speaks for itself.

NASTRAN uses the displacement method and displacements are often more useful in evaluating a problem than stresses so an example of displacement seems in order. Figure 12 shows the Z or axial displacement of grid point number 155 which is at the mid point of the primary bearing surface (on the pitch line) for Heywood loads. In this plot the displacements have been connected to reference the bottom of the fillet as the zero point. The difference here is well defined although not as marked as is the fillet stress case, probably because fillet stress is a much more localized effect than this displacement.

CONCLUSION

In conclusion this paper has attempted to define a method of elastic-plastic analysis of individual thread teeth. The problem of how to define reasonable loading condition for a specific practical problem has not been defined. Even with this limitation, an example has shown the relative magnitude of several loading effects. The reader should pay particular attention to the very definite effect of friction on the behavior of the thread.

CORRECTED
DISPLACEMENT
VS.
SHEAR TRANSFER

FOR THE

BRITISH
STANDARD
BUTTRESS

USING THREE
DIFFERENT
FRICTION
VALUES

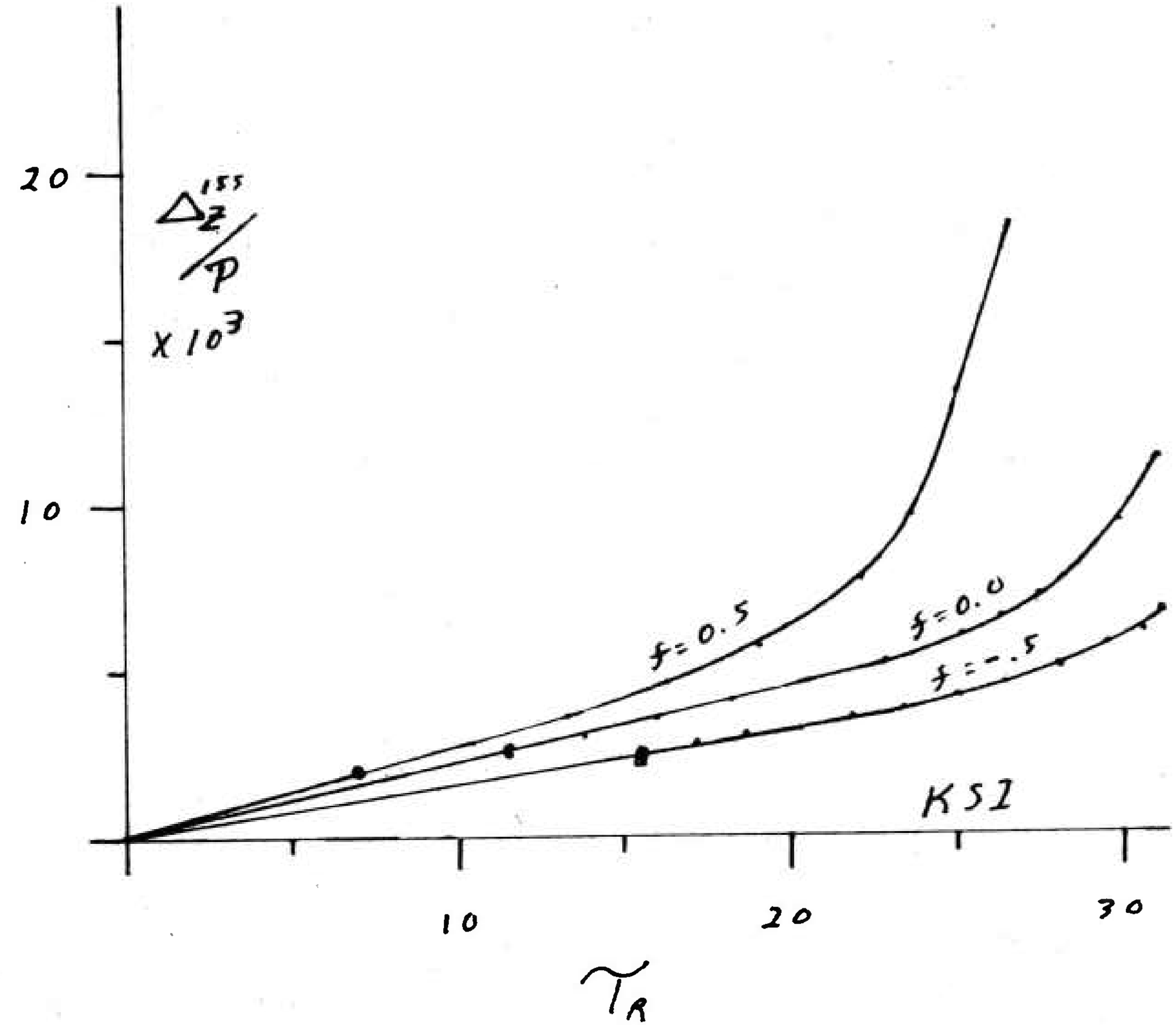

FIGURE 12

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